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AUTHOR(S): J. A. Barclay

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MAGNETIC REFRIGERATION FOR SPACECRAFT SYSTEMS*

J. A. Barclay
Group P-10, MS764
Los Alamos National Laboratory
Los Alamos, New Mexico 87545

ABSTRACT

Magnetic refrigerators, i.e. those that use the magnetocaloric effect of a magnetic working material in a thermodynamic cycle, offer potentially reliable, and efficient refrigeration over a variety of temperature ranges and cooling powers. A descriptive analysis of magnetic refrigeration systems is performed with particular emphasis on more efficient infrared detector cooling. Three types of magnetic refrigerator designs are introduced to illustrate some of the possibilities.

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NOMENCLATURE

| | |
|-------------|---|
| A_c | - contact area, m^2 |
| A_{cs} | - cross section area, m^2 |
| B | - magnetic field, Tesla |
| B_J | - Brillouin function, dimensionless |
| C_B | - heat capacity at constant field, $J/kg\ K$ |
| C_0 | - heat capacity at zero field, $J/kg\ K$ |
| C_p | - heat capacity at constant pressure, $J/kg\ K$ |
| D | - demagnetizing factor, dimensionless |
| G | - molecular weight, $kg/mole$ |
| J | - total electronic angular momentum operator, dimensionless |
| L | - length, m |
| M | - magnetization, $Am^2/mole$ |
| M_s | - saturation magnetization, $Am^2/mole$ |
| N | - Avogadro's number, $mole^{-1}$ |
| P | - pressure, Pa |
| ΔP | - pressure drop, Pa |
| \dot{Q}_c | - cooling power, W |
| \dot{Q}_H | - heat rejection power, W |
| R | - gas constant, $8.13\ J/mole\ K$ |
| S | - entropy, $J/mole\ K$ |
| T | - temperature, K |
| T_C | - cold-bath temperature, K |
| T_H | - hot-bath temperature, K |
| T_{IN} | - inlet temperature, K |
| T_{OUT} | - outlet temperature, K |
| T_0 | - Curie temperature, K |
| ΔT | - magnetocaloric adiabatic temperature change, K |
| \dot{V} | - volume flow rate, m^3/s |
| x | - argument of Brillouin function, dimensionless |

| | |
|-------------|---|
| d | - characteristic channel diameter, m |
| d_p | - particle diameter, m |
| g | - Lande g-factor, dimensionless |
| h | - conductance, W/m^2K |
| k | - Boltzmann constant, J/K |
| v | - velocity, m/s |
| ϕ | - porosity, dimensionless |
| μ_B | - Bohr magneton $J/Tesla$ |
| η | - efficiency, dimensionless |
| κ | - fluid thermal conductivity, W/mK |
| λ_0 | - molecular field constant, dimensionless |
| ν | - viscosity, Pas |
| μ_0 | - permeability constant, Wb/Am |
| ρ | - density, kg/m^3 |
| θ | - solid temperature, K |

INTRODUCTION

The requirement for refrigeration for detectors, instruments, and experiments in spacecraft is increasing because of the need to increase heat load capability and to extend mission life. The specific cooling requirements depend upon the mission and the state of technology available. Table I lists most of the typical spacecraft refrigeration methods that have been successfully flown or are under development.

Refrigeration is a key problem for most developing spacecraft thermal systems because many Watts of cooling must be provided with high reliability, long life-time, and high efficiency. Development of high-reliability, high-efficiency refrigerators has been hindered due to lack of definition, planning and inadequate funding.

TABLE I
SPACECRAFT REFRIGERATION METHODS

| COOLING METHOD | TEMPERATURE RANGE (K) | LOAD/YEAR MISSION |
|--|-----------------------|-------------------|
| Radiant Coolers | 70-100 | 0-10 mW |
| Stored Solid Cryogen Coolers | 10-90 | 0-800 mW |
| Stored Liquid Helium Coolers | 1.5-4.2 | 0-100 mW |
| He-3 Coolers | 0.3 | 0-100 μ W |
| Dilution/Demagnetization Refrigerators | 0.001-0.3 | 0-100 μ W |
| Mechanical Refrigerators | 4-300 | 0-300 W |
| Thermoelectric Refrigeration | | |
| Absorption Refrigerators | | |

In order to see some of the problems faced, consider the following example. Table II lists the features of an advanced infrared (I.R.) detector cooling system. There are no refrigerators in existence that can meet all of these requirements simultaneously.

Table II
I. R. COOLING SYSTEM REQUIREMENTS

| | |
|-------------------|--------------------------------|
| Reliable | Rapid Cool Down |
| Compact | 10 K to 100 K Load Temperature |
| Efficient | Several-Year Mission Lifetime |
| Low Microphonics | Several-Watts Cooling Power |
| Low Thermophonics | |

Also, the reliability and efficiency are becoming increasingly important as the lifetime of the mission and the heat loads increase. If 10 W of refrigeration at 20 K can be reliably provided with 25% of Carnot efficiency instead of 5%, then refrigeration input power can be reduced from 2800 W to 560 W and waste heat removal components can be correspondingly reduced.

The purpose of this paper is to describe the concept of magnetic refrigeration, to discuss the components of a magnetic refrigeration system, and to show that there are possible systems that can provide many of the requirements for refrigeration for spacecraft applications, in particular high efficiency with good prospects for high reliability.

MAGNETIC REFRIGERATION

Magnetocaloric Effect

"Magnetic refrigerators" exploit the temperature and magnetic field dependence of the magnetic entropy of a solid material to extract heat from a low temperature source and transfer it to a higher-temperature sink. The entropy change relevant to these processes is given by

$$dS = \left(\frac{\partial S}{\partial T}\right)_B dT + \left(\frac{\partial S}{\partial B}\right)_T dB = \frac{C_B}{T} dT + \left(\frac{\partial M}{\partial T}\right)_B dB \quad (1)$$

where S is entropy, B is the magnetic field, C_B is heat capacity at constant magnetic field, M is magnetization, and T is absolute temperature. Thus, in order to predict the isothermal entropy change or the adiabatic temperature change with magnetic field variation, the zero-magnetic-field heat capacity C_0 and the equation of state of the magnetization are required. The magnetic-field-dependent heat capacity can be obtained by using

$$C_B = C_0 + T \int_0^B \left(\frac{\partial^2 M}{\partial T^2}\right)_B dB \quad (2)$$

The equation of state for paramagnets and ferromagnets is given by

$$M(B, T) = M_S B_J(x) \quad (3)$$

where M_S is the saturation magnetization and $B_J(x)$ is the Brillouin function.² For a paramagnet

$$x = g\mu_B J/kT \quad (4)$$

where g is the Lande g -factor, μ_B the Bohr magneton, J the total electronic angular momentum operator, and k the Boltzmann constant. For a ferromagnet, in the molecular-field approximation,

$$x = g\mu_B \left[B - (D\mu_0 n/G)M + (\lambda_0 \mu_0 /G)M \right] / kT \quad (5)$$

where D is the demagnetizing factor, μ_0 is the permeability constant, n is the density, G is the molecular weight, and λ_0 the molecular field constant given by

$$\lambda_0 = 3kT_0 G / N\mu_0 g^2 \mu_B^2 J(J+1) \quad (6)$$

with T_0 equal to the Curie temperature. Substitution of Eqs. (5) and (6) into (3) gives a transcendental equation for M , which can be solved by iteration. Paramagnets at low temperature (1-20 K) and ferromagnets near their Curie temperatures (20-300 K) show appreciable entropy changes with field; values range from $\geq 1R$ at low temperatures to 0.1-0.2 R at room temperature, with a maximum of $R/(2J+1)$, where R is the gas constant. Magnetic materials have adiabatic temperature changes of 10-20 K in 6-10 T fields.

In order to fully appreciate the capability of magnetic systems, it is instructive to compare the simple thermodynamics of magnetic and gas systems because many scientists and engineers are more familiar with gas systems. The corresponding variables in the two systems are P , T , and B , T where P is pressure. The entropy equation for magnetic solids is given by Eq. (1) and the corresponding gas equation³ (for an ideal gas) is

$$dS = \left(\frac{\partial S}{\partial T}\right)_P dT + \left(\frac{\partial S}{\partial P}\right)_T dP = \frac{C_P}{T} dT - \left(\frac{R}{P}\right) dP \quad (7)$$

where C_P is the heat capacity at constant pressure.

Equations (7) and (1), (3), (5) can be used to calculate the isothermal entropy changes and adiabatic temperature changes under corresponding P or B changes. Figure 1 presents the entropy-temperature diagram for a pressure ratio of 10:1 and a magnetic field change of 10 T. Further comparisons show that

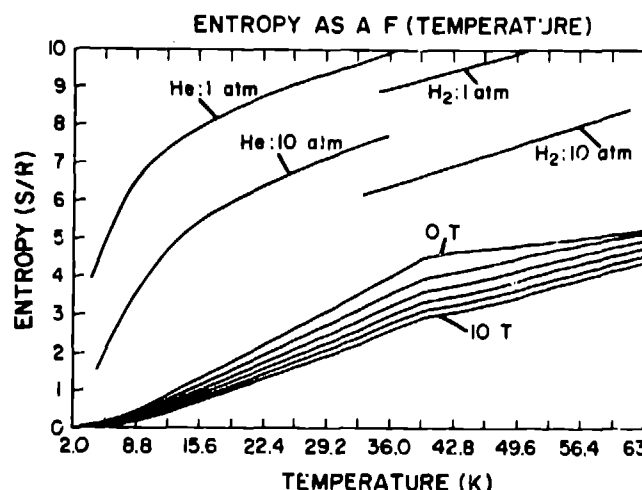


Fig. 1. Entropy as a function of temperature for a magnetic solid, hydrogen and helium gas. The gas compression ratio is 10:1 and the Curie temperature of the solid is 40 K with a field change of 10 T.

magnetic refrigerants have larger entropy change per unit volume, even at room temperature, because of their higher density, but that gases offer a much larger temperature change above 30-40 K. Magnetic refrigerants have an approximately constant adiabatic temperature change ΔT for all temperatures while gases have a ΔT that increases as the temperature increases. The large ΔT upon compression of a gas leads to large inefficiency in a compressor when a reasonable through-put is required. The magnetic system has a much smaller ΔT which leads to good efficiency, but may require a penalty in the size of the room-temperature heat exchanger.

Thermodynamic Cycles for Magnetic Refrigerators

In order to use the magnetocaloric effect in a refrigerator a suitable thermodynamic cycle must be executed. Several magnetic analogues of gas cycles exist. The historic Carnot cycle, two isothermal steps and two adiabatic steps, is easy to execute in a magnetic system. Consider a ferromagnetic material near its Curie temperature such that the material can be isolated from or put in contact with hot and cold baths at will. The first stage of this cycle is an isothermal magnetization while in contact with the hot bath; the heat of magnetization is rejected into the hot bath. Next, an isolated (adiabatic), partial demagnetization is performed which cools the material to a lower temperature. The third step puts the material in contact with the cold bath as the demagnetization is continued to zero field; heat is drawn from the cold bath. The final step of the cycle is an adiabatic partial magnetization back to the original hot

temperature. These two isothermal steps and two adiabatic steps constitute a magnetic Carnot cycle. The temperature span of a Carnot cycle is limited to 5-10 K with an ~ 10 -T field change, but no regeneration is required. Since larger temperature spans are generally required, other cycles must be used. The magnetic Brayton cycle consists of two adiabatic steps and two isofield steps. It requires regeneration but can cover much larger temperature spans than a Carnot cycle. The Brayton cycle is very attractive because of the natural way of coupling to the external heat exchangers through the temperature change caused by the adiabatic field changes. Magnetic Ericsson and Stirling cycles are also possible. Ericsson cycles consist of two isothermal steps and two isofield steps. Stirling cycles require two isothermal steps and two isomagnetization steps. Both of these cycles require regeneration but can span large temperature differences. Because these cycles require excellent heat transfer between the heat-exchange fluid and the source and sink to obtain the isothermal steps, the Brayton cycle tends to be easier to implement in a practical device.

A Magnetic Refrigeration System

In order to integrate a magnetic refrigerator with a complete spacecraft system, the refrigerator subsystem components and their constraints must be fully understood. Table III lists the characteristics of a magnetic refrigerator system.

TABLE III

THE CHARACTERISTICS OF A MAGNETIC REFRIGERATOR SYSTEM

PRINCIPLE

- o Magnetocaloric Effect
- o Adiabatic Temperature Change of 10-20 K in 6-10 T
- o Isothermal Entropy Change of 1R(4K) to 0.1R(300K)

SYSTEM COMPONENTS

- o Solid Magnetic Working Material
- o Heat-Exchange Fluid
- o Fluid-Drive Pump
- o External Heat Exchangers
- o Magnetic-Material Drive Motor
- o Superconducting Magnet
- o Superinsulated Dewar

Let us now consider each component in more detail in an attempt to understand what characteristics spacecraft magnetic refrigerators might have.

Magnetic Material. As shown in Fig. 1, the entropy change caused by the magnetic field decreases as the operating temperature deviates from the Curie temperature. A temperature span of 40-60 K can be achieved with a single material; but if a larger temperature span is required, a series of materials must be used so that each material operates near (and through) its Curie temperature. For example, if a temperature span of 20 K to 200 K was desired for a particular spacecraft application, at least three materials would be required, each spanning about 60 K. This requirement is no problem since there are many gadolinium-based compounds that have Curie temperatures ranging from 20 K to 300 K. Table IV lists some of these compounds. Ideally, a continuous range of Curie temperatures would be used;

TABLE IV
GADOLINIUM COMPOUNDS OR ALLOYS AND THEIR CURIE TEMPERATURES⁴

| Compound/Alloy | T ₀ (K) | Compound/Alloy | T ₀ (K) |
|---|--------------------|--|--------------------|
| Gd ₂ .71S ₄ | 21 | Gd _{0.5} Ag _{0.5} (amorphous) | 100 |
| GdAg _{0.8} In _{0.2} | 24 | GdAg _{0.5} In _{0.5} | 111 |
| Gd ₂ .73S ₄ | 28 | Gd _{0.68} Mo _{0.32} | 125 |
| Gd _{0.65} Th _{0.35} Al ₂ | 30 | Gd _{0.8} Au _{0.2} (amorphous) | 150 |
| Gd Ni ₅ | 32 | Gd Al ₂ | 153 |
| Gd Ni ₂ (amorphous) | 38 | Gd _{0.7} La _{0.3} | 185 |
| Gd ₂ .276S ₄ | 42 | GdGa | 200 |
| Gd Ag _{0.3} Al _{0.7} | 50 | Gd _{0.67} Y _{0.33} | 211 |
| Gd Ag _{0.7} In _{0.3} | 57 | Gd _{0.85} In _{0.15} | 224 |
| Gd ₂ .8S ₄ | 58 | Gd _{0.80} Y _{0.20} | 254 |
| Gd N | 65 | Gd Zn | 268 |
| Gd Ni ₂ (crystalline) | 81 | Gd _{0.90} Y _{0.10} | 281 |
| Gd _{0.8} Th _{0.2} Al ₂ | 90 | Gd | 293 |
| | | GdMn ₂ | 300 |

more research is required before the best selection of materials can be made, but there should be no problem obtaining suitable ones.

Heat-Exchange Fluid. A heat exchange fluid must be used to couple the magnetic solid to the hot and cold external heat exchangers and to effect the regeneration. If relatively small temperature spans are required, various cryogenic liquids could be used: but, generally, supercritical helium, hydrogen, or neon can be used to span from low temperatures to room temperature. If the design uses fluid as the regenerator, care must be taken to avoid mixing across the temperature gradient.^{5,6} It is better to use the magnetic solid as the regenerator as well as the working material to avoid excess entropy production due to fluid mixing. Heat transfer in the regenerative parts of the cycle must be excellent to attain high efficiencies. Excellent heat transfer capability can be obtained using porous-beds. The conductance between a flowing gas and the particles of the bed has been studied by several workers.^{7,8,9} One of the more successful correlations for porous beds at room temperature is that of Coppage and London⁹, given by

$$h = 0.21(\rho v C_p / a) \left(\frac{2avd_p}{\mu(1-a)} \right)^{-0.31} \left(\frac{C_p \mu}{k} \right)^{-1} \quad (8)$$

where h is the conductance, a is the bed porosity, d_p the bed particle diameter, and v the superficial fluid velocity (as if the bed were empty); ρ , C_p , μ , and k are the fluid density, heat capacity, viscosity, and thermal conductivity, respectively. Once a fluid is chosen, the bed parameters d_p , a , and, indirectly, v , can be varied to obtain as large an hA_c as possible, where A_c is the bed contact area given by

$$A_c = \frac{6(1-a)A_{cs}L}{d_p} \quad (9)$$

Here A_{cs} is the cross sectional area and L is the length of the bed. Because hA_c can be made very large in porous beds, the heat transfer can be extremely good. However, it is not possible to make the heat transfer arbitrarily high because the flow losses become intolerable.

If the temperature span is 20-300 K, the fluid must be either helium or hydrogen gas. Passage of gas through the porous magnetic solid results in a pressure drop in the gas and, hence, heat generation. The most general equation for the pressure drop through a porous bed is Ergun's equation¹⁰,

$$\Delta P = \left[\frac{150\mu(1-a)}{d_p^2 v} + 1.75 \right] \frac{(1-a)Lv^2}{a^3 d_p} \quad (10)$$

where ΔP is the pressure drop through a bed of length L . The term in brackets is called the friction factor and contains the viscous-flow term (first term) and the turbulent flow term (1.75). When Eqs. (9) and (10) are considered together, we see that although a decrease in d_p increases A_c , it also increases ΔP . At some point, the flow loss ΔP , where \dot{V} is the volume flow rate, produces so much irreversible entropy that the efficiency drops to zero. However, it is possible to choose a value of d_p such that the heat transfer is excellent and the flow losses are minimal.

External Heat Exchangers. The detector-module power dissipation along with radiative and conductive heat leaks provide the low temperature thermal load for the refrigerator. There may be other thermal loads at higher temperatures from thermal shields, etc. All of these loads plus the magnetic work and heat due to losses must be rejected as heat at some higher temperature, e.g., by radiation at >200 K. An external heat exchanger performs the function of connecting the heat-exchange fluid (and, indirectly, the working material) to the thermal loads and rejection sink. One simple type of heat exchanger is a channel through a metal block attached to the load or radiator. For the steady state case, neglecting viscous losses, energy conservation gives the following differential equation:

$$\rho v C_p \frac{dT}{dx} = \frac{4h}{d}(a-T) \quad (11)$$

where C_p is the heat-exchange fluid heat capacity, h the conductance between the block at temperature a and the fluid at temperature T , and d is the characteristic channel diameter. Assuming that a is

essentially constant in steady state, Eq. (11) can be solved to yield

$$T_{OUT} = \theta + (T_{IN} - \theta) \exp\left(\frac{-4hL}{\rho v C_p d}\right), \quad (12)$$

where T_{IN} is the heat exchanger fluid at the inlet of the block and T_{OUT} is the temperature after length L . Equation (12) shows that the design parameters for a given fluid are v and L/d . The mass flow rate is determined by the required cooling power and $T_{OUT}-T_{IN}$.

The temperature difference, $T_{OUT}-T_{IN}$, must also be equal to ΔT , the adiabatic temperature change for the magnetic material. The mass flow ρv is determined by picking the diameter of the channel, which in turn determines h , which determines L/d from Eq. (12). For example, if we want 1 W of cooling power at $\theta = 20$ K and the heat exchange fluid is 1.0 MPa helium gas, then, for $\Delta = 10$ K, \dot{m} becomes 2×10^{-5} kg/s, ρv is 25 kg/m²s, h becomes 700 W/m²K, and L/d is 83, so L equals 0.83 m. The corresponding pressure drop in this channel is 250 Pa, i.e., completely negligible. A corresponding calculation at a rejection temperature of 200 K gives similar results. The conclusion of this exercise is that the heat exchange can be easily accomplished at the expense of a slightly larger temperature span, e.g., for 20 K source and 200 K sink, $T_C = 8$ K and $T_H = 216$ K. Counter-flow heat exchange would be even easier.

Magnetic Material Drive Motor. The work for the magnetic refrigeration cycle has to be provided by a drive motor of some type, possibly including a magnetic heat engine in analogy to a gas Vuilleumier cycle. In a wheel-type design, this work can be introduced by a torque on the drive shaft. A reciprocating-type design requires a force compensation mechanism because the magnetic work is produced by the difference between two large magnetic forces, according to $(\mathbf{M} \cdot \nabla) \mathbf{B}$, as the magnetic material enters and leaves the field. It is also possible to consider moving the magnet instead of the magnetic material, which illustrates an additional degree of design flexibility that magnetic systems have, i.e., the intensive variable B can be sculptured much more than the corresponding variable P in gas systems.

Superconducting Magnet. The need for a high-field magnet is definitely the biggest disadvantage to magnetic refrigeration. The effects of magnetic field on other parts of a spacecraft, e.g., magnetometers and IR detectors, would have to be carefully determined. Of course, one saving feature is that magnetic fields can be effectively shielded but only at the expense of high permeability or superconducting shielding materials. However, if the reliability and efficiency are high, this shielding penalty may not be serious.

Superconducting magnet technology using NbTi and Nb₃Sn materials is well developed. Magnets capable of producing a 10-T field are available using either material; we have taken 10 T as the practical upper limit for superconducting magnets in this application. The magnetic work in the thermo-dynamic cycle is not put into the refrigerator by charging the magnet, so that once the field is established the magnet can operate in the persistent mode. The leads to the magnet can then be removed to reduce the heat leak into the helium. The steady magnetic field eliminates charging losses due to flux jumping and will reduce eddy current

losses to a negligible level. The magnet will require an initial liquid helium transfer, but thereafter the refrigeration requirement can be provided by a small magnetic refrigerator operating from, for example, 20 K to 4 K which will add a comparatively negligible parasitic heat load at 20 K. For example, in a 20 W at 20 K refrigerator, calculations indicate that an additional 120 mW load is required at 20 K.

Superinsulated Dewars. Modern helium, fiber-glass Dewars with many layers of vapor-cooled shielding and superinsulation are readily available and have remarkably low losses. Boil-off rates of 20-30 h/l are not uncommon. This boil-off translates into a thermal influx of a few mW. This type of Dewar can also maintain its integrity for years if properly designed with gettering material, such as activated charcoal.

Costs. Spacecraft-worthy magnetic refrigerators have not yet been designed, so costing is difficult. However, in general, it can be stated that none of the components in a magnetic refrigerator system are very costly, although a complete system can only be properly costed after a design is available.

POSSIBLE DESIGNS

Wheel Type

The wheel concept¹² illustrated in Fig. 2 (taken from Ref. 12) provides continuous refrigeration but has a limited temperature span, as can be seen from the S-T curves of Fig. 1. A span of 40-70K

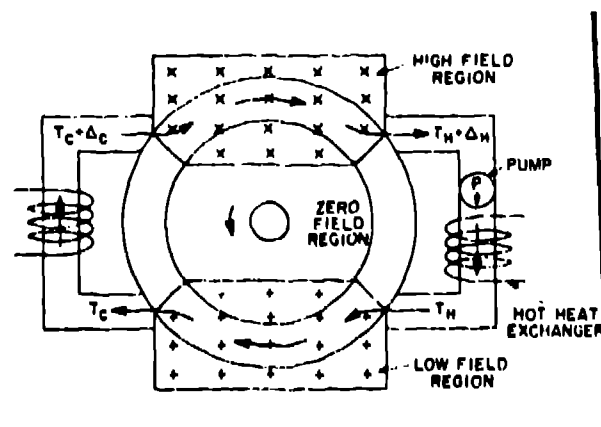


Fig. 2. The magnetic wheel concept illustrated by a magnetic material in the rim of a wheel executing a Brayton cycle.

is possible; but the entropy change decreases as the span increases, so 40 K may be a practical limit. If an IR-detector refrigerator with a load at 20 K and a radiative sink at 200 K were desired using the wheel concept, the overall temperature span of ~200 K could be obtained from 4 or 5 magnetic wheels in series. Reference 12 shows that a Brayton-cycle wheel spanning ~40 K should be able

to reach 70% of Carnot, which would give ~ 17% of Carnot for 5 stages extending from ~ 10 K to ~ 215 K. This design should work but is probably not the best way to span large temperature ranges. There are reports on three magnetic-wheel refrigerators available.^{13,14,15}

Reciprocating Type

The basic concept of reciprocating designs is illustrated in Fig. 3, taken from Ref. 5. This

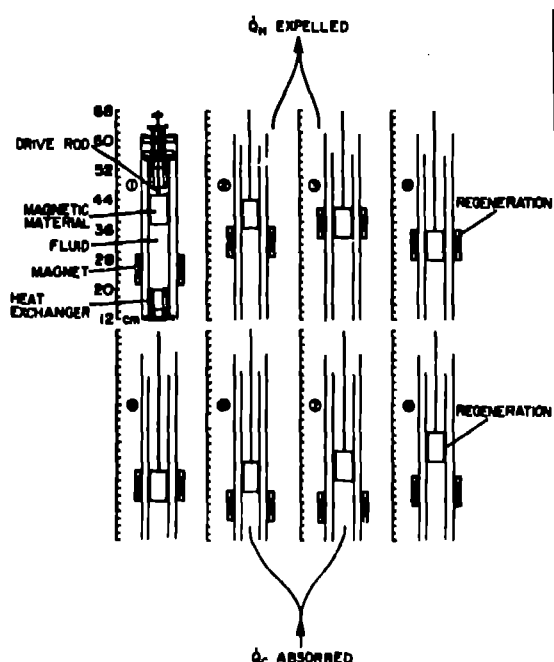


Fig. 3. The magnetic reciprocating concept illustrated by an eight-step sequence executing a magnetic Ericsson cycle.

design type was the first one suggested for use above 1 K.¹⁷ The temperature span is again about 40-70 K, but the refrigeration is not completely continuous. However, the magnet can be a simple solenoid and entrained fluid does not cause problems. The regenerator losses can be serious, particularly since mixing of fluid along the temperature gradient can rapidly produce entropy. If the mixing problem can be solved, staged reciprocating refrigerators should be possible. There are two reports on reciprocating magnetic refrigerators available in addition to Ref. 17.^{5,6}

Active Magnetic Regenerator Type

Many of the limitations associated with the previous two designs can be avoided or eliminated by the use of a new concept called magnetic regenerative refrigeration.

The active magnetic regenerator is a device composed of several magnetic materials that are thermodynamically cycled to provide the refrigeration over an extended temperature range. The basic theory is that of an ordinary regenerator except that the temperature of the materials can be changed by the application or removal of a magnetic field and that a thermal wavefront propagates back and forth in the regenerator.

Each different material executes a small Brayton cycle near its Curie temperature; but when all of the materials are combined, they yield a Brayton cycle over an extended temperature range, e.g., 10-220 K. The basic cycle is illustrated in Fig. 4 and is described as follows. Consider a

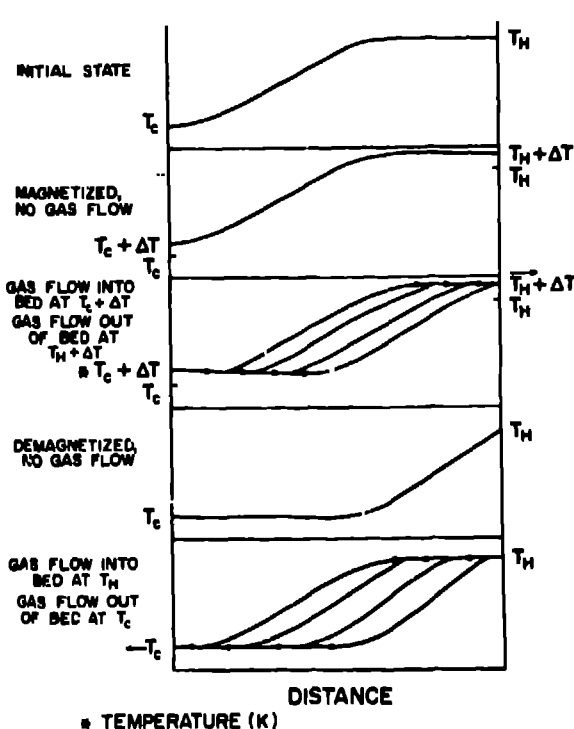


Fig. 4. The active magnetic regenerator concept illustrated by a five-part sequence of the temperature profile along a regenerator executing a magnetic Brayton cycle over an extended temperature span.

porous-bed regenerator composed of a series of different ferromagnetic materials with Curie temperatures T_C gradually decreasing from the sink temperature T_H to the load temperature T_C . Also, consider that the temperature gradient is nearly uniform but displaced to the left of the center in the regenerator, as shown in the top frame of Fig. 4. (For start up from a warm condition, i.e., T_H everywhere, it takes several cycles to reach the condition assumed above; so, for simplicity, we start with the temperature gradient established.) Upon application of a magnetic field, the temperature all along the bed will adiabatically increase by ΔT which is about 15-20 K for a 10-T field. (One of the characteristics of magnetic refrigerants is that ΔT is roughly independent of T if the material is near its Curie temperature.) After the field is applied, helium or hydrogen gas, at temperature $T_C + \Delta T$ is pushed through the bed from the cold end, which is now at $T_C + \Delta T$. As the gas at $T_C + \Delta T$ enters the bed, the gas will warm as the bed cools, and a thermal wavefront of magnitude $T_H - T_C$ will be established, as shown in the middle frame of Fig. 4. The overall wavefront will propagate through the regenerator (to the right

in the middle frame of Fig. 4) as gas continues to flow into the bed at $T_C + \Delta T$. The gas leaves the regenerator at $T_H + \Delta T$ until the thermal wavefront arrives at the hot end of the regenerator, at which time the temperature of the exiting gas drops to T_H . When this happens, the gas flow is stopped and the regenerator is adiabatically demagnetized. The temperatures all along the bed drop

by ΔT as shown in frame 4 of Fig. 4, in preparation for the reverse flow of gas. The gas that came out of the regenerator at $T_H + \Delta T$ during the magnetized stage is put through a heat exchanger and cooled to T_H before it is pushed back into the regenerator after demagnetization. Another thermal wave of magnitude $T_H - T_C$ is established; but it travels in the opposite direction to the first thermal wavefront, as shown in the bottom frame of Fig. 4.

The gas exits the cold end at T_C and is then put in contact with the load. When the gas temperature at the cold end of the regenerator increases from T_C to $T_C + \Delta T$, the gas flow is stopped and the cycle is repeated as the regenerator is again magnetized.

A key point in this concept needs to be emphasized. The finite heat transfer between the heat-exchange gas and the magnetic solid, along with the axial conduction in the bed, will tend to disperse the thermal wave front as it propagates through the regenerator. In a non-active (non-magnetic-normal type) regenerator these effects will gradually reduce any thermal gradient by spreading the temperature change out across the whole bed. However, in an active magnetic regenerator, the wavefront propagates back and forth through the regenerator without complete dispersion because the wavefront is sharpened every half-cycle by distributed refrigeration. This concept is still in the development stage, but preliminary experiments¹⁸ and calculations¹⁹ indicate that it can provide refrigeration over large temperature spans, e.g., 10-220 K, at very impressive efficiencies, ~ 50% of Carnot. This concept has also been embodied into a wheel and a reciprocating design, but no detailed design work has been completed.

DISCUSSION

Magnetic refrigeration has been introduced for IR sensor cooling but may have many other space applications. The key feature is its potential higher efficiency; but preliminary calculations also indicate that the volume and mass of a magnetic regenerative refrigerator will be less than comparable gas refrigerators. The mass calculations included mu-metal shieldings around the entire refrigerator dewar to eliminate the effects of stray magnetic fields on other spacecraft components. The reliability of a magnetic refrigerator is uncertain because of a need for seals, bearings, motors, etc., but the cycle frequency is very low, i.e., ~ 0.1 Hz (6 RPM) so there is a good possibility high reliability might be achievable. The need for 4.2 K refrigeration suggests that the first applications should be in the 4-20 K temperature range.

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